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# Calculation Method of Pipeline Vibration with Damping Supports Made of the MR Material

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**Abstract**

Wire damping materials (such as Metal Rubber, MR, or "spring cushion", etc.) are widely used for protection of a pipeline against vibration. A calculation method for an arbitrary shaped pipeline with damping supports made of the MR material was developed by means of the finite element ANSYS software package. Dependencies of MR support stiffness and energy dissipation coefficient on various parameters (material density, wire diameter, thickness, preliminary static deformation) are obtained as these two characteristics influence the quality of MP supports. An equivalence viscous damping coefficient was obtained on the basis of the equality of the hysteretic loop area for the viscous damping system and dry friction in the wire material. The method takes into account a non-linearity of the MR material by the iteration method. The method allows obtaining a pipeline vibration amplitude and stress in the pipeline for its 3D vibration. Calculation results are proved by experiment with the ARAMIS non-contact measurement system. The error is less than 9%, thus, it is possible to use this method for practical application: placement of pipeline supports and design of these supports parameters. It allows reducing a large experimental work of stress measurement in pipelines during pipeline system design. Any engineer who is able to work with the ANSYS software can use it. It is possible to develop this method for calculation of a pipeline system with many pipelines connected by damping supports.


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## 1. Introduction

Vibration of pipeline is a reason of pipeline breaking. It determines a life-time of pipelines. Dry friction damping supports are used widely for protection of pipelines against vibration. These supports use very often a damping material made of pressing wire, such as MR (Metal Rubber) [1, 2]. MR material is manufactured by cold pressing of wire spiral. It has high damping, high strength, high ability to work in aggressive media. Analogous wire materials are “metal-flex”, “spring cushion” [3, 4], “wire mesh damper” [5]. However MR material (as other pressing wire materials) is non-linear [6], its stiffness and energy dissipation coefficient depend on deformation amplitude and preliminary static deformation. A method for calculation of pipeline vibration should take into account this nonlinearity. However in the present time the calculation methods of pipeline vibration are linear most of all [7-10]. These methods don’t take into account a complexity of pipeline shape, they consider the pipeline as a straight or curved beam. Maximal complexity now is a pipeline as some branches [11]. These methods take into account a damping in a liquid into pipeline but not in supports. On this reason these methods usually allows to obtain resonance frequencies of pipeline but not stress, or give a relative distribution of stress but not an exact stress value [12]. The aim of the present research is to develop a method for calculation of space vibrations of arbitrary shape pipeline, allowing calculating the stress. This method will be a basement for pipeline support design.

## 2. Methods

The basement of the present method is ANSYS software. Finite element Pipe16 was used for pipelene, finite element Combin14 – for supports, finite element Mass21 – for concentrated mass. The element Combin14 allows using a viscous damping, however MR material has dry friction. Equivalence viscous damping coefficient was obtained for equality of hysteretic loop area for viscous damping system and dry friction system. This coefficient is $C_v = \psi C / (2\pi\omega_0)$ (here $\omega_0$ is resonance frequency).

The calculation method needs support characteristics: stiffness $C$ and energy dissipation coefficient $\psi$ . It is possible to find it for MR material (as any other dry friction system) from hysteretic loop under static load in coordinates “deformation x – load P” (Figure 1).

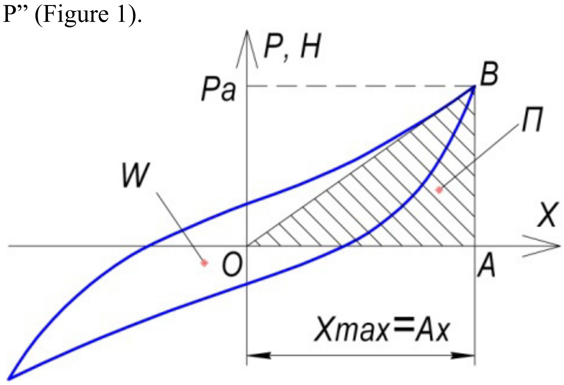


Fig. 1. Hysteretic loop of dry friction system.

Characteristics of stiffness and damping of MR are:

- middle-cycle stiffness $C_{cp}=P_a/A$ (here $P_a$ is amplitude of static load, $A$ – amplitude of displacement);
- energy dissipation coefficient $\psi = W / \Pi$ , (here $W$ is hysteretic loop area, it is equal to a dissipated energy during one deformation cycle, $\Pi$ is potential energy of deformation, it is equal to an area of triangle OAB (Figure 1) [13]).

It is supposed in a present paper that static and dynamic characteristics of dry friction systems are equal. It is proved by similarity of dynamic experiment and dynamic characteristics of dry friction system calculated by static experiment [14 - 16].

Characteristics of support were obtained by results of static experiment with thin plates made of MR material with different thickness $H$ (from 1.15 to 3.45 mm), density ρ (from 1.115 to 2.028 g/sm$^3$), relative preliminary static deformation $\varepsilon_Q = Q / H$ (from 0.087 to 0.268; here $Q$ is absolute preliminary static deformation), wire diameter $d_w$ (from 0.09 to 0.12 mm) and relative amplitude of deformation $\varepsilon_A = A / H$. A range of $\varepsilon_A$ changing was taken from 0.043 to 0.146. If this value is more, life-time of support will be too little [17].

Equation for $C$ and $\psi$ are [18]:

$$C = 14(1+363\varepsilon_A^{3.5})\ (1+727\varepsilon_Q^{3.45})\ d_w^{1.2} H^{-0.33}\ (\varepsilon_Q / 0.262)^{0.063H}\, S\ (\overline{\rho} / 0.2)^{1.1+0.01H/d_w} \qquad (1)$$

$$\psi = 1.71(1-7.23H^{0.13}(\varepsilon_A - 0.048))\ (1-0.1(10.5\varepsilon_Q - 1)^2)\ d_w^{0.138} H^{0.15}\ (0.095/\varepsilon_Q)^{0.057H} \times \times (0.2/\overline{\rho})^{0.37+0.015H/d_w} \qquad (2)$$

Here $\overline{\rho} = m_{MR} / (\pi H r_0^2 \rho_S)$ is relative density; it is equal to relationship of MR material element mass $m_{MR}$ to element made of steel with density $\rho_S$ and the same volume.

MR material is non-linear. Values of $C$ and $\psi$ depends on relative amplitude of dynamic deformation $\varepsilon_A$ which depends on $C$ and $\psi$. To take into account these dependencies an iteration method was used. By equations (1) and (2) initial values of stiffness and energy dissipation coefficient $C^{(0)}$ and $\psi^{(0)}$ were calculated for initial deformation amplitude $A^{(0)} = 0.1$ mm. After it amplitudes of vibration in the place of damping support in vertical and horizontal directions $A_V^{(1)}$ and $A_H^{(1)}$ are obtained in harmonic calculation by ANSYS. By these values $A_V^{(1)}$ and $A_H^{(1)}$ more precise values of $C^{(1)}$ and $\psi^{(1)}$ were calculated. Next ANSYS calculation allows to obtain next approximation $A_V^{(2)}$, $A_H^{(2)}$, $C^{(2)}$ and $\psi^{(2)}$. This process repeats till $A_V^{(N)} \approx A_V^{(N-1)}$ and $A_H^{(N)} \approx A_H^{(N-1)}$ within the limits of required error.

As example of pipeline was chosen a pipe made of titanium alloy. Its outer diameter is 6.2 mm, inner diameter 4.0 mm, its shape is complex enough to apply results for this pipe for any pipeline of arbitrary shape (Figure 2). Both of pipe ends have connections similar to connections of pipeline with units of engine. There is a support in the middle of pipe. Its wideness is 11 mm, it is possible to place in it elements made of MR material with different parameters and different preliminary static deformation Q. All supports have stiff connection with a beam, the beam is mounted on vibration stand.

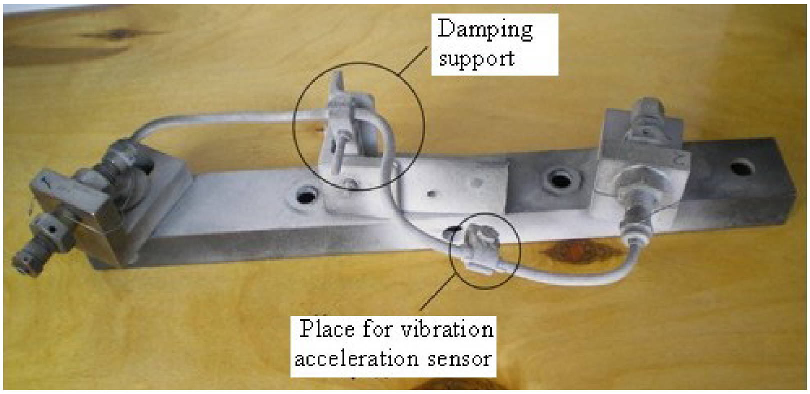


Fig. 2. A researched pipeline.

Vibration stand VDS-1500 was used for experimental research of pipeline vibration. For vibration measurement a non-contact ARAMIS system was used. This system includes two high-speed cameras which takes pictures of object simultaneously. Comparison of these two pictures allows obtaining an object displacement and deformation. To check ARAMIS measurement a usual vibration acceleration sensor was used. Difference of resonance frequency by ARAMIS and vibration acceleration sensor is less than 2%.

Frequency and amplitude of pipeline vibration depend on stiffness and energy dissipation in connections of pipeline with units of engine too. An energy dissipation coefficient in connections was obtained by vibration amplification coefficient $\eta$ on a resonance for a pipeline without damping support. By experiment $\eta$=46, thus for linear approximation $\psi_S \approx 2\pi / \eta = 0.14$ .

Stiffness of connection for displacement was obtained experimentally by applying a force to connection. Its value is about $C_0$ = 1111 N/mm. Stiffness of connection for rotation was taken as infinitely large.

### 3. Results

Comparison of experiment and calculation for MR element with $H$ = 2.1 mm, relative static deformation $\varepsilon_Q$ = 0.14 and relative density of MR material $\bar{\rho}$ = 0.25 is given in Tables 1 and 2. Amplitudes of pipeline vibration in a place of damping support are presented on Figure 3. The method allows calculating of stress in the pipeline too. For example an equivalent stress in the pipeline at resonance frequency 207 Hz is presented at Figure 4.

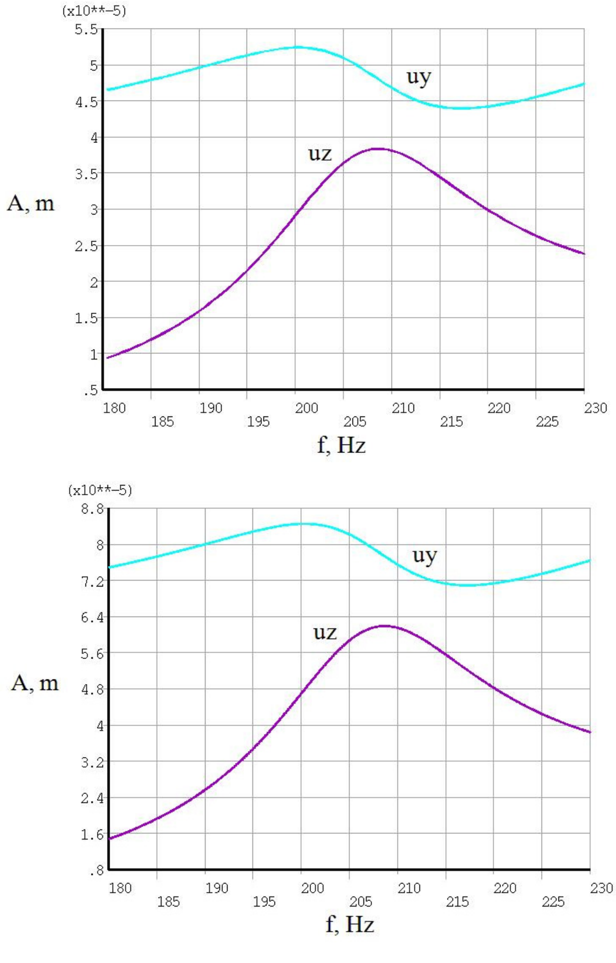


Fig.3. Amplitude – frequency dependencies ($u_y$ – vertical direction, $u_z$ – horizontal direction).
First picture $W_1$ = 30 m/s$^2$, second picture $W_1$=50 m/s$^2$.

Table 1. Resonance frequencies of pipeline.

| Vibration stand input acceleration $W_1$, m/s$^2$ | 30 | 50 |
|---|---|---|
| Resonance frequency by ARAMIS measurement, Hz | 195.1 | 193.5 |
| Resonance frequency by ANSYS calculation, Hz | 207 | 207 |
| Error, % | 5.8 | 6.7 |
| Vertical vibration amplitude, mm | 0.044 | 0.071 |

Table 2. Amplitudes of pipeline vibration in a place of damping support.

| Vibration stand input acceleration $W_1$, m/s$^2$ | 30 | 50 |
|---|---|---|
| Resonance frequency, Hz | 206.6 | 206.6 |
| Amplitudes of pipeline vibration, mm | Vertical/ Horizontal | Vertical/ Horizontal |
| ARAMIS measurement | 0.052/0.04 | 0.074/0.065 |
| ANSYS calculation | 0.05/0.037 | 0.08/0.059 |
| Error, % | 4/7.5 | 8/9 |

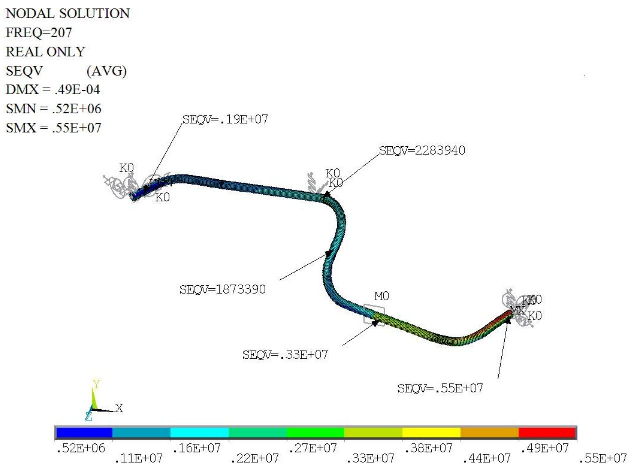


Fig. 4. An equivalent stress in the pipeline at resonance frequency 207 Hz. $W_1$ = 30 m/s$^2$

## 4. Discussion

It is possible to see from Tables 1 and 2 that displacement calculation error is 4…9% near damping support. Frequency calculation error is about 6.5%. It is satisfactory for practice because a difference of MR material parameters is about 10% on technology reasons [1].

The calculated value of displacement near damping support allows correcting deformation amplitude of MR material, and stiffness and energy dissipation coefficient respectively. Thus it is possible to understand a vibration condition of pipeline and to obtain optimal characteristics of MR material for support (density, preliminary static deformation etc).

3D vibration of pipeline with complex shape has some peculiarities. For example, it is possible that a maximal displacement amplitude is in other direction than direction of excite vibration [12]. The present method allows obtaining this picture.

Because the pipeline moves not only in the direction of excite vibration, the damping element should encircle a pipe as a ring. Many of pipeline supports in present time have gaps between damping elements, it means it design isn't right.

## 5. Conclusions

Thus the developed method of calculation of space vibration of pipeline with damping supports made of MR material allows obtaining resonance frequencies of arbitrary shape pipeline, displacement of its points and stress in the pipeline for any amplitude of harmonic vibration load. It means it is possible to use the presented method to reduce a large experimental work of measurement a stress in pipelines during pipeline systems design. This method takes into account a nonlinearity of MR material, dependency of its properties on deformation amplitude. Previous methods considered a pipeline only as a composition of linear and bended parts and used linear properties of damping materials. The presented method is simple enough to use it by usual engineer who is able to work with ANSYS software.

The presented method doesn't take into account a vibration of liquid into pipeline. If it is necessary one should to supplement this method with other calculation method [9].

A pipeline has not only branches [11], but connection with other pipelines very often. In this case there is an influence of its vibration on each other. It is possible to develop the present method to take into account this influence: to model the same way another pipelines and damping connections between them.

The presented method allows taking into account parameters of pipeline connections with units of engine, however a future researches of stiffness and damping in these connections are required.

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